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FLOW BOILING OF ETHANOL IN SMALL DIAMETER TUBES

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ABSTRACT

Flow boiling occurs where the liquid is naturally circulated or is pumped through a heated channel, typically the tube. In flow boiling the liquid-vapor phase-change processes play an important role in many technological applications,
Presented in the paper data partially explain the complex structure of two-phase laminar flow of ethanol in small diameter tubes. Accomplished were systematic investigations of flow boiling of ethanol in two tube dimensions, i.e. 1.15mm and 2.3mm. Utilised have been modern thermovisual techniques for measurements of wall temperature. Identified was the „M" shape distribution of heat transfer coefficient. This work partly explains the specific features of two-phase flow and heat transfer inside minichannels.

INTRODUCTION

Boiling heat transfer, as one of the most efficient techniques for removing high heat fluxes, has been studied and applied in practice for a very long time. Despite numerous applications the theoretical approaches to modeling of flow boiling still require substantial progress as determination of heat transfer and pressure losses is done mostly by means of empirical correlations. Their drawback is that they feature fluid-dependent coefficients and are not general. Such correlations must be therefore used with a special care and precaution, only in the range of conditions such correlations have been developed for. That fact also disables direct application of correlations developed for conventional channels to small diameter channels. Recently there is also progress attained using the structure dependent modeling using the flow maps, which is however tedious and not very convenient for engineering applications.

In the paper presented is a study into flow boiling of ethanol in two small diameter silver tubes with inner diameters of 1.15mm and 2.3mm, respectively. The experiments have been accomplished for a wide range of quality variation, mass flow rate and heat fluxes, i.e. x=0 - 0.9, G=100-700 kg/m^{2}s and q= 50-300 kW/m^2, respectively. The saturation temperature ranged from 50 to 85^{o}C. The objective of the present study was to shed more light into the problem of flow boiling in small diameter tubes, as during experiments it turned out that the obtained distributions of heat transfer distribution with respect to quality were following in some cases the M-shaped distribution of heat transfer coefficient detected earlier by Bar-Cohen and Rahim (2009) [1] and Mikielewicz and Klugmann (2010) [2]. The ethanol has been selected as a test fluid and experiments were accomplished primarily in the laminar range of flow conditions.

NOMENCLATURE

c_p	J/kg K	specific heat
Con		Constraint number, Con=$(\sigma/g/(\rho_l-\rho_v))^{0.5}/d$
d	m	channel inner diameter
f_1, f_{1z}	[-]	functions
G	kg/m^{2}s	mass flowrate
J_G	[-]	dimensionless vapour velocity
k_L	W/mK	liquid thermal conductivity
l	m	channel length
p	Pa	pressure
Pr	[-]	Prandtl number
R_{MS}	[-]	two-phase flow multiplier due to Müller-Steinhagen and Heck
Re	[-]	Reynolds number
T	K	temperature
x	[-]	quality
X_{tt}	[-]	Martinelli parameter
g	m/s^2	gravity
u,w	m/s	velocity components
α	W/m^{2}K	heat transfer coefficient
μ	Pa s	dynamic viscosity
ρ	kg/m^3	density
δ	m	liquid film thickness
σ	N/m	surface tension
τ	Pa	shear stress

Subscripts

l	liquid
LO	liquid only
TPB	two-phase boiling
TPK	two-phase condensation

EXPERIMENTAL FACILITY

The experimental rig has been designed and constructed by Klugmann (2009) [3] and modified later by Glinski (2010) [4] as a compact, highly integrated mobile unit, fig. 1, suitable for experiments with flow boiling and dryout. The principal part of the research rig is a closed loop of a working fluid.

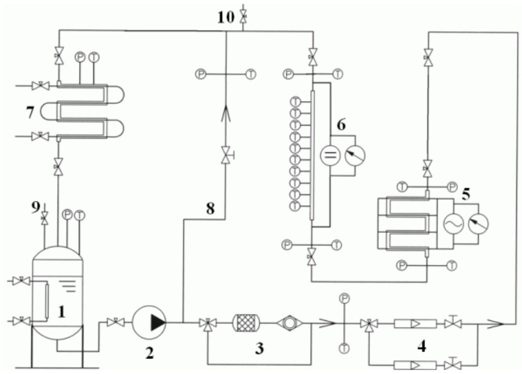

Figure 1 Schematic diagram of experimental facility: 1– main tank; 2 – pump section; 3 – filter/dryer; 4 – mass flowmeters; 5 –pre-heater; 6 – test section; 7 – condenser; 8 – by-pass; 9 – filling valve; 10 – service valve

The flow of working fluid is forced by a set of two electrically-powered pumps, composed in series, capable to deliver the mass flow rate up to 200 kg/h and overpressure up to 8 bars. Gear pumps have been chosen to limit any arising flow pulsations. Adjustment of the mass flow is realized by changing voltage of the pump's power supply or using the by-pass. Working medium is pumped from the main tank through the Danfoss mass flow meter MASS D1 3 type working with MASS 6000 19" IP20 interface. Such system, gives about 0.3% measurement accuracy. In the present work the mass flow range from 1 to 4.5 kg/h has been considered. Then the working fluid goes to the pre-heater, where it attains required input parameters. Isobaric pre-heating is realized in the stainless steel tube powered by low voltage, high current DC power supply. Such arrangement gives up to 1.2 kW of heating power. In this way, the full range of quality x is possible to be achieved at the test section input. Current, voltage, inlet and outlet temperatures and pressure are measured on the pre-heater to determine a corresponding heat flux and quality x from the appropriate heat balance. From the pre-heater the medium goes directly to the test section. In this experiment a silver tube of 1.15mm or 2.3mm inner diameter, and the length of 38cm was used. Current, voltage, inlet temperature, inlet pressure, outlet temperature and outlet pressure are measured at the test section to determine the corresponding heat flux, subcooled liquid temperature, saturation temperature of boiling liquid and pressure drop.

The working medium flows into the tube with a pre-defined quality x and is heated further to get the expected boiling conditions. Heating in the test section is realized using a low voltage, high current DC power supply and can be adjusted from 0 up to 1 kW of heating power, which returns a heat flux up to 364 kW/m^2. Current, voltage, inlet temperature, inlet pressure, outlet temperature and outlet pressure are measured at the test section to determine the corresponding heat flux, subcooled liquid temperature, saturation temperature of boiling liquid dryout and pressure drop. All the data are collected automatically using PC computer with a data acquisition interface. The measuring system uses specially developed in-house software TERMOLAB 06, Klugmann (2009). From the test section the medium goes to the water cooled condenser and back to the main storage tank. The unit is also equipped with the filter/dryer and an additional pre-heater in the main tank (using hot water).

RESULTS OF EXPERIMENTS

The objective of that part of work was to collect information about laminar heat transfer in flow boiling of ethanol in minichannels (d_h < 3 mm). Range of parameters used in investigations of ethanol flow boiling was: tube diameter D=1.15, 2.3mm, mass velocity G=100 ÷ 700 kg/m^{2}s, heat flux q=50 ÷ 300 kW/m^2, saturation temperature T_{sat}=50 ÷ 85^{o}C, quality x=0-1, tube length z=380mm.

In flow boiling in conventional size channels there is observed a maximum in heat transfer coefficient distribution in function of quality which is found at about x≈0.8. That location corresponds to the existence of annular flow structure in the tube. In case of small diameter channels that maximum moves towards smaller values of quality, Bar-Cohen and Rahim [1]. In case of channel diameters equal to 2.3mm that maximum was found for x≈0.4 and in case of even smaller diameters at x≈0.1, [4]. That has also been confirmed in research accomplished by Thome (2007) [5], who confirms that the bubbly flow is found in small diameter channels only up to qualities not exceeding x≈0.1. In studies of flows in minichannels there can also exist an additional maximum of heat transfer coefficient in its distribution in function of quality, see fig. 2. The only known study thus far, where such situation was noticed in the paper [1]. Similar findings were also found in the recent study in [2,3].

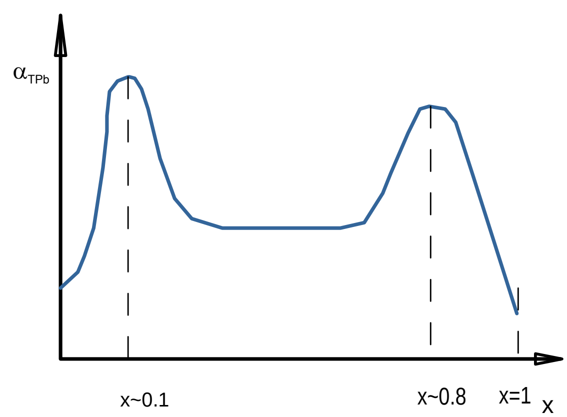

Figure 2 Schematic of M-shape distribution of heat transfer coefficient minichannels

The mechanism of development of M-shape distribution can be explained in the following way. Values of heat transfer coefficient gradually increase from the values which are obtained at small subcoolings to the local maximum close to x=0. Then, the decrease is observed followed by a plateau and subsequently by a slight increase of heat transfer coefficient. At values of quality about 60% the second maximum is observed. After that another decrease of heat transfer coefficient is found. The smallest values are reached when x approaches 1, which is generally related to the dryout conditions.

The M-shape distribution of heat transfer coefficient seems to be a specific phenomenon devoted merely to two-phase flows in minichannels. Initiation of boiling on the wall and related to it acceleration of saturated liquid flow induces significant rise of heat transfer coefficient, significantly above the level of single phase values. That may explain the first steep rise on the heat transfer coefficient distribution. Subsequent transition to the slug/plug flow structure renders gradual decrease of heat transfer coefficient, as the vapour slugs form conditions for local evaporation of thin liquid film, separating the liquid and the wall, and even to development of dry patches, which impair heat transfer. The transition to annular flow structure again results in increase of heat transfer coefficient, as a result of development of evaporating thin liquid film on the wall. The evaporating film thinners and splits into rivulets causing local dryout conditions. It is very rare, if possible, to find in literature distributions of heat transfer coefficient in a whole range of quality variation. Usually the experimental data refer to a range of quality and enable to reflect only a part of the M-shaped curve. A detailed analysis of experimental data enables to conclude that different sets of data describe exactly different parts of M-shape distributions. That is the reason why in some cases authors are claiming the reduction of heat transfer coefficient with quality, Kandlikar (2001) [6], and sometimes the increasing trend, Huo et al. (2007) [7]. The M-shape distribution of heat transfer coefficient explains that behaviour, [3].

In the course of the study the developed earlier model for heat transfer coefficient was used:

In case of consideration of bubble generation the following expression is valid for calculation of heat transfer, Mikielewicz et al. [8,9]:

$$\frac{\alpha_{TPB}}{\alpha_l} = \sqrt{R_{MS}^n + \frac{1}{1+P}\left(\frac{\alpha_{PB}}{\alpha_l}\right)^2} \qquad (1)$$

In equation (1) the term, $P=2.53\times10^{-3}Re^{1.17}Bo^{0.6}(R_{MS}-1)^{-0.65}$, has been established by a method of multiple regression fitting. The pool boiling heat transfer coefficient α_{PB}, is to be calculated from the relation due to Cooper [10]. The applied heat flux is incorporated through the boiling number Bo, defined as, $Bo=q/(Gh_{lv})$. For the same difference between the wall and saturation temperature there is a different temperature gradient in the fluid in case of pool boiling and flow boiling. In the case of flow boiling the boundary layer is thinner and hence the gradient of temperature is more pronounced, which suppresses generation of bubbles in flow boiling. That is the reason why heat flux is included in modeling. That term is more important for conventional size tubes, but cannot be totally neglected in small diameter tubes in the bubbly flow regime, where it is important. Postulated form of correction has an appearance preventing it from assuming values greater than one, which was a fundamental weakness of the model in earlier modifications.

In the form applicable to conventional and small diameter channels the Muller-Steinhagen and Heck [11] model yields:

$$R_{MS} = \left[1 + 2\left(\frac{1}{f_1} - 1\right)x\,Con^m\right]\cdot(1-x)^{1/3} + x^3\frac{1}{f_{1z}} \qquad (2)$$

where $Con=(\sigma/g/(\rho_l-\rho_v))^{0.5}/d$ and m=0 for conventional channels. Best consistency with experimental data, in case of small diameter and minichannels, is obtained for m=-1. In equation (2) $f_1=(\rho_l/\rho_v)\ (\mu_l/\mu_v)^{0.25}$ for turbulent flow and $f_1=(\rho_l/\rho_v)(\mu_l/\mu_v)$ for laminar flows. Introduction of the function f_{1z}, expressing the ratio of heat transfer coefficient for liquid only flow to the heat transfer coefficient for gas only flow, is to meet the limiting conditions, i.e. for x=0 the correlation should reduce to a value of heat transfer coefficient for liquid, $\alpha_{TPB}=\alpha_l$ whereas for x=1, approximately that for vapour, i.e. $\alpha_{TPB}\cong\alpha_v$. Hence $f_{1z}=\alpha_v/\alpha_l$, where $f_{1z}=(\lambda_v/\lambda_l)$ for laminar flows and for turbulent flows $f_{1z}=(\mu_v/\mu_l)(\lambda_l/\lambda_v)^{1.5}(c_{pl}/c_{pv})$. The correlation (1) seems to be quite general, as confirmed for example by the recent study by Chiou et al. [12].

In fig. 3 presented are results of measurements accomplished at the Heat Technology Department of Gdansk University of Technology and compared with the predictions obtained using the model (1) [9]. Over 80% of data fall into the error band of ±30% and 95% within ±50%. That error is evenly distributed for all qualities, fig. 4. In case of small qualities, the consistency between measured data and calculations performed using the model described by equation (1) is very good. The consistency slightly decreases for greater qualities. Analysis of figures 5 to 8 shows that the heat transfer coefficient depends also on the value of heat flux, at constant value of mass velocity.

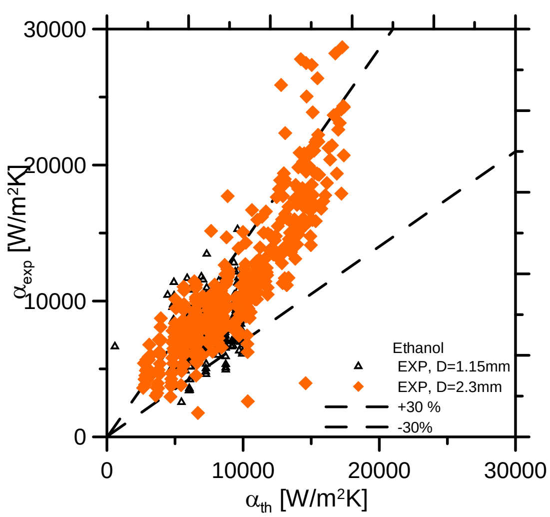

Figure 3 Experimental heat transfer coefficient vz. heat transfer coefficient calculated using (1), d=1.15mm and 2.3mm

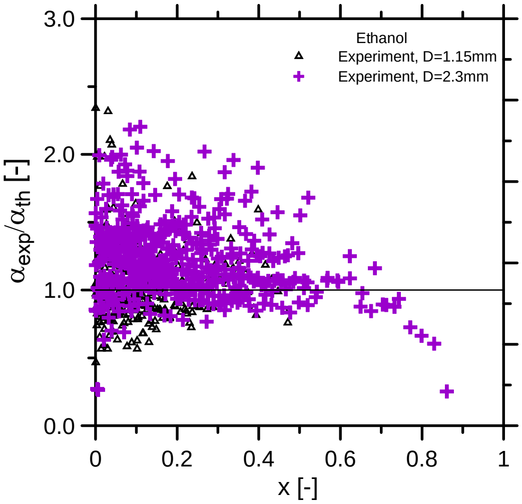

Figure 4 Ratio of experimental heat transfer coefficient to theoretical one calculated using (1) in function of quality, d=1.15mm and 2.3mm

The experimental discrepancy is related to the fact that there could be small pressure pulsations in operation of the pumps. Also a strongly non-equilibrium character of considered processes could influence the readings.

There ought to be noticed, however, the presence of two maxima of heat transfer coefficient, one for x ≈0.1÷0.3 and the one for x≈0.6÷0.7, which is consistent with findings of Klugmann (2009) [3] for R123 and Glinski (2010) [4] for R134a and SES36. In the present work it was not possible to obtain the whole range of heat transfer coefficient variation with respect to quality. Only parts of quality range were possible to be obtained, mainly for lower qualities. The maximum corresponding to smaller qualities shifts towards higher values of vapour content with increasing heat flux. There was a significant difficulty in obtaining the systematic data for higher qualities. Some data showing the maximum of heat transfer coefficient at higher qualities was collected and especially can be seen in fig. 9-10.

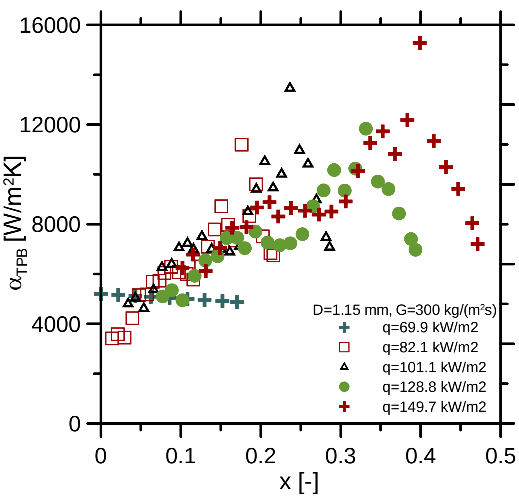

Figure 5 Influence of heat flux on local heat transfer coefficient in function of quality, D=1.15mm, G=300 kg/(m^{2}s)

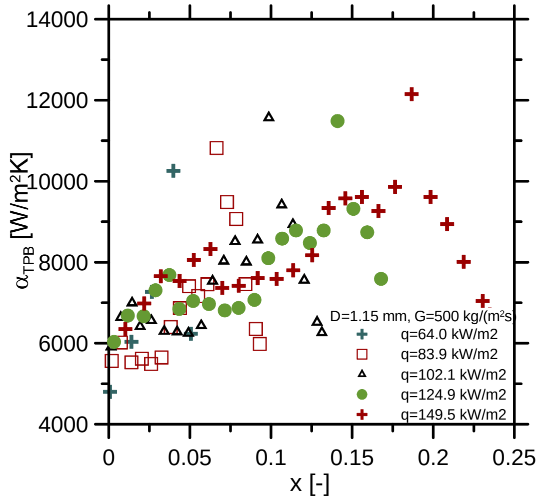

Figure 6 Influence of heat flux on local heat transfer coefficient in function of quality, D=1.15mm, G=500 kg/(m^{2}s)

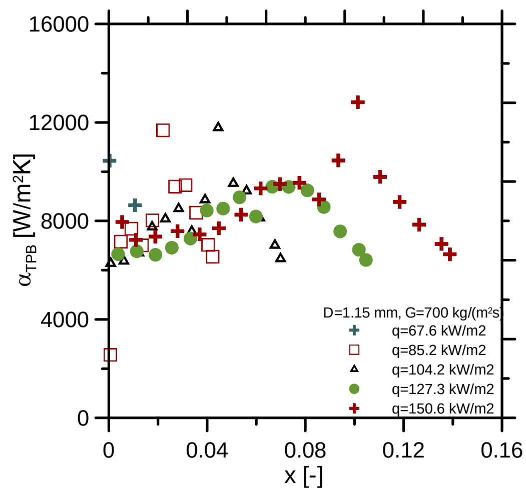

Figure 7 Influence of heat flux on local heat transfer coefficient in function of quality, D=1.15mm, G=700 kg/(m^{2}s)

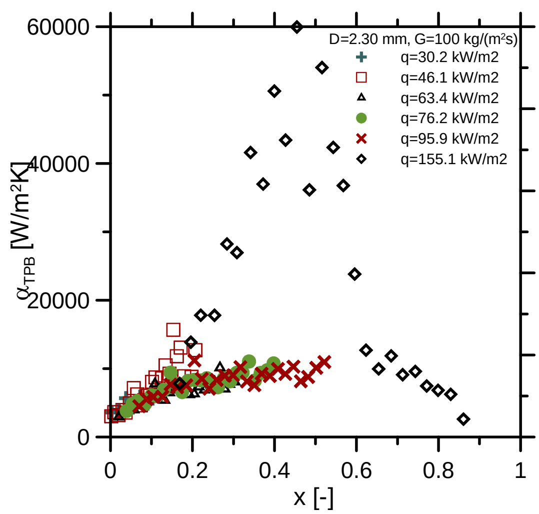

Figure 8 Influence of heat flux on local heat transfer coefficient in function of quality, D=2.3mm, G=100 kg/(m^{2}s)

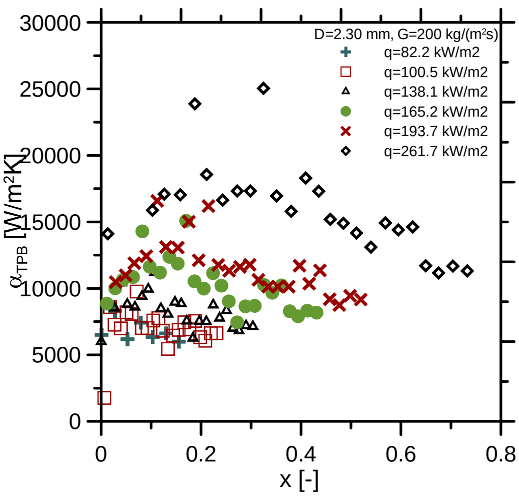

Figure 9 Dependence of heat flux on local heat transfer coefficient in function of quality, D=2.3mm, G=200 kg/(m^2s)

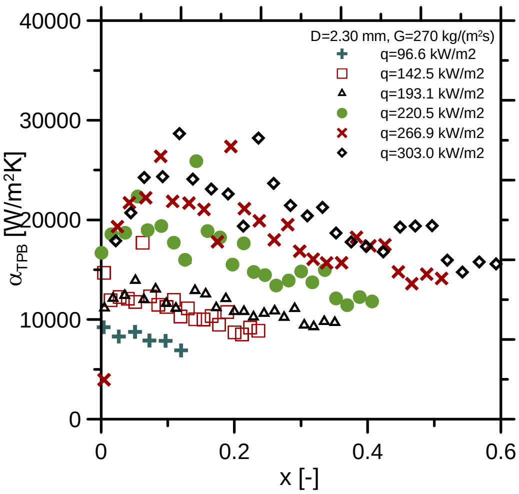

Figure 10 Dependence of heat flux on local heat transfer coefficient in function of quality, D=2.3mm, G=270 kg/(m^2s)

It can be noticed that the presented picture of the influence of heat flux at a constant value of mass velocity is not monotonic for values of quality. For small qualities (x<0.4) it can be seen that with the increase of heat flux the heat transfer coefficient increases. On the other hand for greater values of quality the reverse trend is observed, which points at the existence of the second local maximum, which is in line with the investigations of Bar-Cohen and Rahim (2009). After the second maximum all distributions indicate that the heat transfer coefficient decreases with increase of quality.

COMPARISONS WITH OTHER AUTHORS DATA

In literature there is a number of correlations for description of saturated flow boiling. For consideration here three correlations have been taken, which have specifically been developed for flow boiling in minichannels. These are correlations due to Steinke and Kandlikar (2004) [13], Lazarek and Black (1982) [14], and Owhaib (2007) [15]. The experimental results

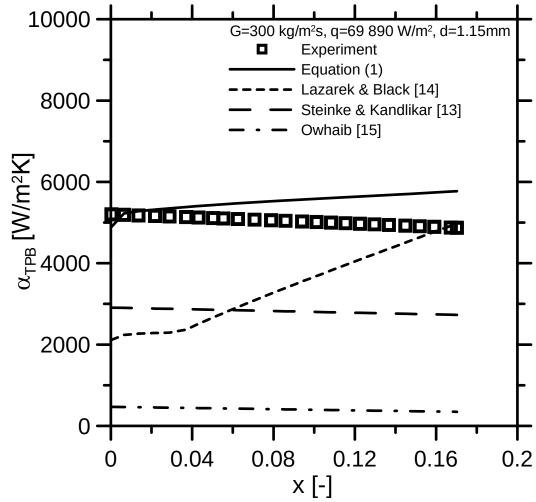

Figure 11 Comparison of different correlation with measurement data in function of quality, G=300 kg/m^2s, q=69 930 W/m^2, D=1.15mm

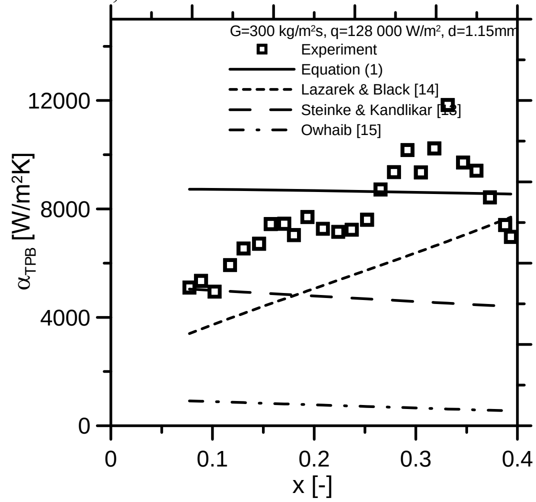

Figure 12 Comparison of different correlation with measurement data in function of quality, G=300 kg/m^2s, q=128 000 W/m^2, D=1.15mm

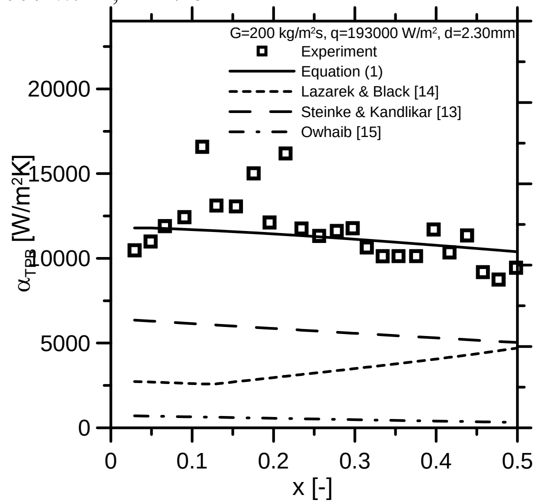

Figure 13 Comparison of different correlation with measurement data in function of quality, G=200 kg/m^2s, q=193 000 W/m^2, D=2.3mm

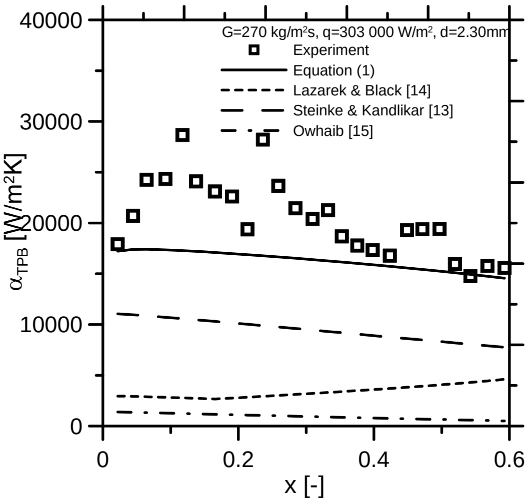

Figure 14 Comparison of different correlation with measurement data in function of quality, G=200 kg/m2s, q=261 000W/m2, D=2.30mm

have also been also compared against the model, equation (1). The results of comparisons have been presented in fig. 11–13.

The absolute superiority of the model due to Mikielewicz (2009) [9] can be seen from the attached figures. It can be said that only in selected cases other correlations perform to a satisfactory extent. The correlation (1) can describe the increasing trend of heat transfer coefficient as well as the decreasing one, whereas the other ones solely the either one. In case of the model (2.10) there is however a discrepancy for larger qualities, which means that the correlation describing the flow resistance, in the present case the model due to Muller-Steinhagen and Heck (1986) is not accurate enough and other flow resistance correlation should be sought.

CONCLUSIONS

Presented in the paper partially explain the complex structure of two-phase laminar flow of ethanol in small diameter tubes. Accomplished were systematic investigations of flow boiling of ethanol in two tube dimensions, i.e. 1.15mm and 2.3mm. Identified was the „M" shape distribution of heat transfer coefficient, similarly and in earlier studies by Glinski (2010) for SES36, R134a and R123. Qualitatively and quantitatively different patterns of heat transfer coefficient in function of quality were detected. In a general case two maximum values of heat transfer coefficient as a function of quality can be found. The first one occurs at the small vapour content in the flow in minichannel (quality of about 0.1). In most other authors experiments the decreasing heat transfer coefficient character was noticed because most of the measurements were executed for $0.1 < x < 0.7$, which is before attaining the second heat transfer coefficient maximum conditions. That maximum was noticed for some flow parameters (for bigger qualities, $x \approx 0.7 - 0.8$). Presence of these two maxima leads to the „M-shape" distribution of heat transfer coefficient. This observation was identified only recently in the world literature [10]. Heat transfer coefficient increase with decreasing tube diameter was also shown in the some quality range. Also noticed was the fact, that heat transfer coefficient maximum values are bigger for the smaller channel diameters.

Comparison experimental data with some correlations from literature show that the correlation due to Mikielewicz at al. [13] returns best consistency of those selected for comparisons here. Other correlations are satisfactory only in selected cases.

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REFERENCES

[1] Bar-Cohen A., Rahim E., Modeling and prediction of two-phase microgap channel heat transfer characteristics, Heat Transfer Engineering, 30 (8), pp. 601-625, 2009.
[2] Mikielewicz D., Klugmann M. Wajs J., Experimental investigation of M-shape heat transfer coefficient distribution of R123 flow boiling in small diameter tubes, accepted for publication in Heat Transfer Engineering, 2010.
[3] Klugmann M., Heat transfer intensification in flow boiling in small diameter channels, PhD Thesis, Gdansk Univ., 2009.
[4] Gliński M., Investigations and modeling of critical heat flux in small diameter channels, PhD Thesis, Gdansk Univ., 2010.
[5] Thome J.R. (2007), Chapter I, Wolverine Engineering Databook III, at www.wlv.com/products.
[6] Kandlikar S.G., Fundamental issues related to flow boiling in minichannels and microchannels, Proc. Exp. Heat Transfer, Fluid Mechanics and Thermodynamics, 129-146, Thesalloniki 2001.
[7] Huo, X., Chen, L., Tian, Y.S., Karayiannis, T.G., Flow boiling and flow regimes in small diameter tubes, Applied Thermal Engineering, Vol. 24 pp. 1225-1239, 2007.
[8] Mikielewicz D., Mikielewicz J., Tesmar J., Improved semi-empirical method for determination of heat transfer coefficient in flow boiling in conventional and small diameter tubes, Int. J. of Heat and Mass Transfer, 50, 3949-3956, 2007.
[9] Mikielewicz D., A New Method for Determination of Flow Boiling Heat Transfer Coefficient in Conventional-Diameter Channels and Minichannels, Heat Transfer Engineering, Volume 31 (4), pp. 276-284, 2009.
[10] Cooper M. G., Saturation nucleate pool boiling: a simple correlation, Int. Chem. Eng. Symp.1, 86, pp. 785-793, 1984.
[11] Müller-Steinhagen R., Heck K., A simple friction pressure drop correlation for two-phase flow in pipes, Chem. Eng. Progress, 20, 1986.
[12] Chiou C.B., Lu D.C., Liao C.Y., Su Y.Y., Experimental study of forced convective boiling for non-azeotropic refrigerant mixtures R-22/R-124 in horizontal smooth tube, *Applied Thermal Engineering*, vol. 29, pp. 1864–1871, 2009.
[13] Steinke M, Kandlikar S., An experimental investigation of flow boiling characteristics of water in parallel microchannels, Journal of Heat Transfer, 126 (4), pp. 518-526, 2004.
[14] Lazarek G.M., Black S.H., Evaporative heat transfer, pressure drop and critical heat flux in a small vertical tube with R-113, Int. J. Heat Mass Transfer , Vol. 25, No. 7, pp. 945-960, 1982.
[15] Owhaib W., Experimental heat transfer, pressure drop and flow visualization of R134a in vertical mini/micro tubes, PhD Thesis, KTH, Stockholm, 2007.